

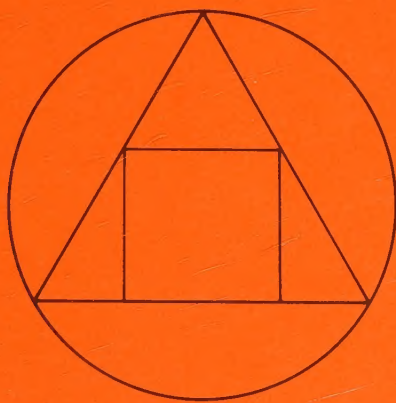
Doctor's thesis

MACHINE ELEMENTS

DIVISION

Lund Technical University

Lund, Sweden



On Hydrodynamic Lip Seals

Bert Andersson

Lund 1973

ON HYDRODYNAMIC LIP SEALS

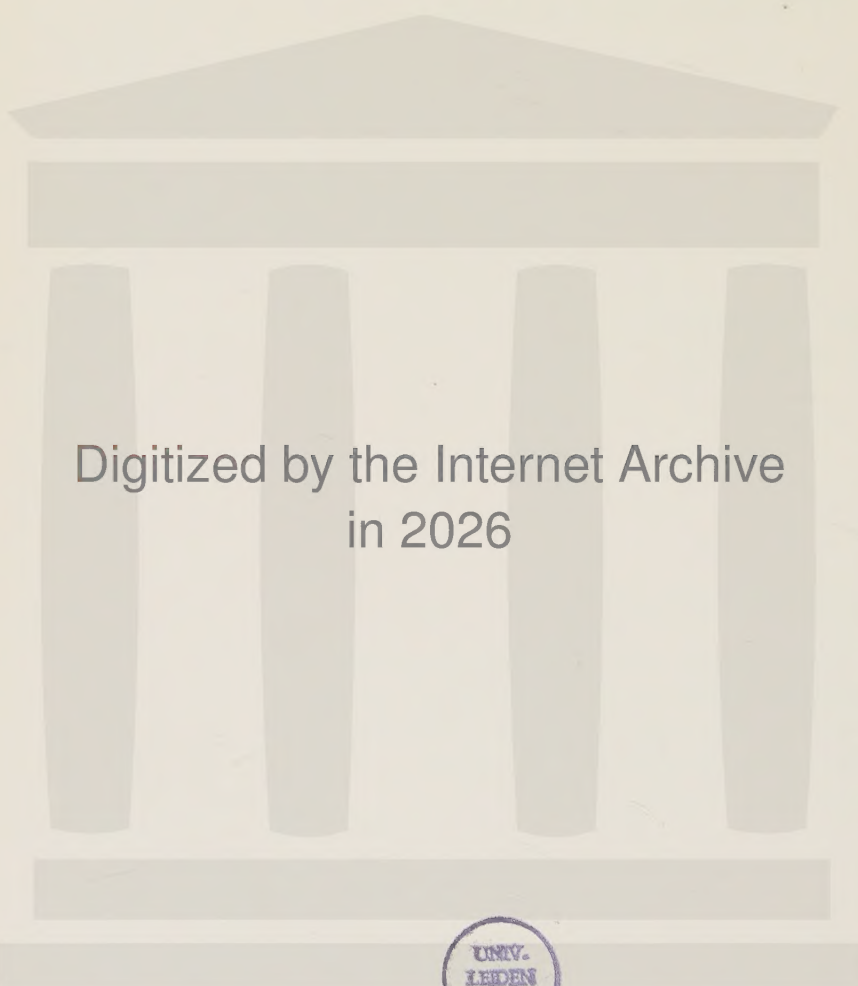
Av

BERT ANDERSSON

Tekn. Lic., VRML

AKADEMISK AVHANDLING

som för avläggande av teknisk doktorsexamen
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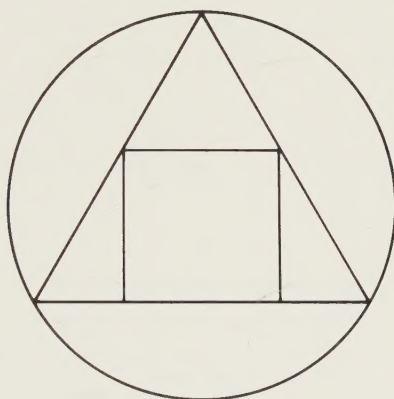
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The following papers are included in this dissertation:

1. Andersson, Bert: On Optimum Design of Hydrodynamic Lip Seals. Acta Polytechnica Scandinavica, Mechanical Engineering Series No. 56, Stockholm 1971.
2. Andersson, Bert: Calculation of Contact Force for Hydrodynamic Lip Seals Including Optimization with Regard to Cross Section of the Lip. Trans. Machine Elements Division, Lund Technical University, Lund 1973.



Optimum Geometry of the Contact

The liquid to be sealed is assumed to be Newtonian and incompressible and the viscosity variations within the film are neglected. Subatmospheric pressures are also neglected.

The geometry of the contact surfaces for the unidirectional and bidirectional seals are shown in figs. 1 and 2.

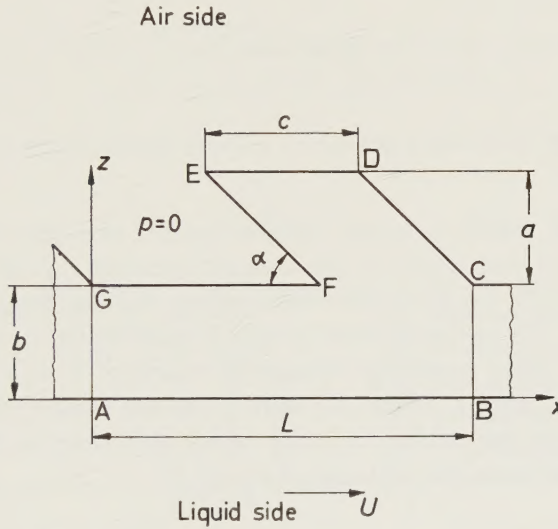


Fig. 1. Contact geometry for the unidirectional seal

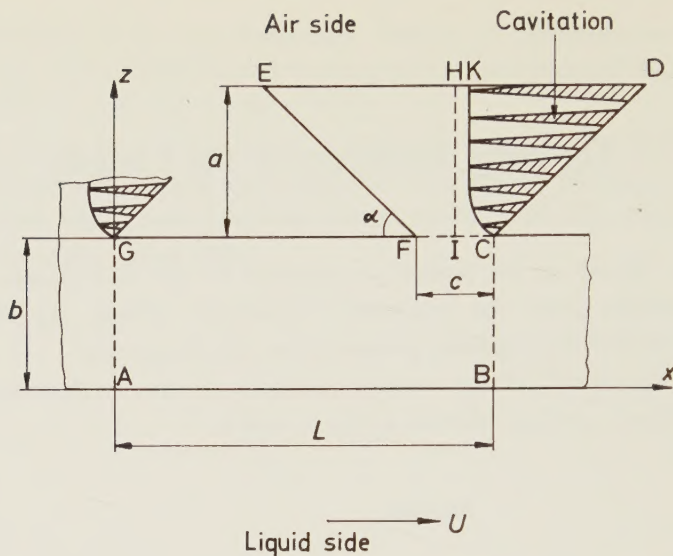


Fig. 2. Contact geometry for the bidirectional seal

The area ABCG is a part of the contact between the static lip and the shaft and CDEF is the contact between one beam and the shaft. The static lip gives the static sealing and the beams the dynamic sealing. An optimum film profile is used which means that the beams have great pumping capacity in relation to their power loss.

In figs. 3 and 4, where pressure fields are shown, the values of Δp_0 represent the sealing capacity which is defined as the pressure difference at which the seal starts to leak.

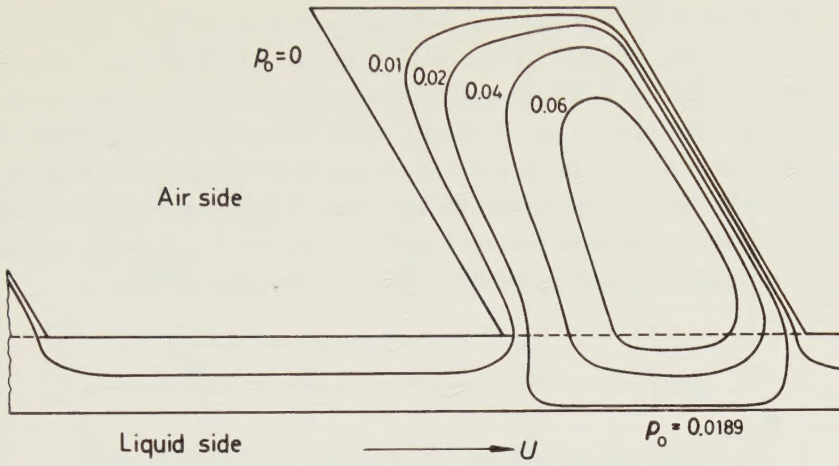


Fig. 3. Pressure field for a unidirectional seal with $a_o=0.433$, $b_o=0.1$, $c_o=0.4$, and $\alpha=60^\circ$. $\Delta p_o=0.0189$.

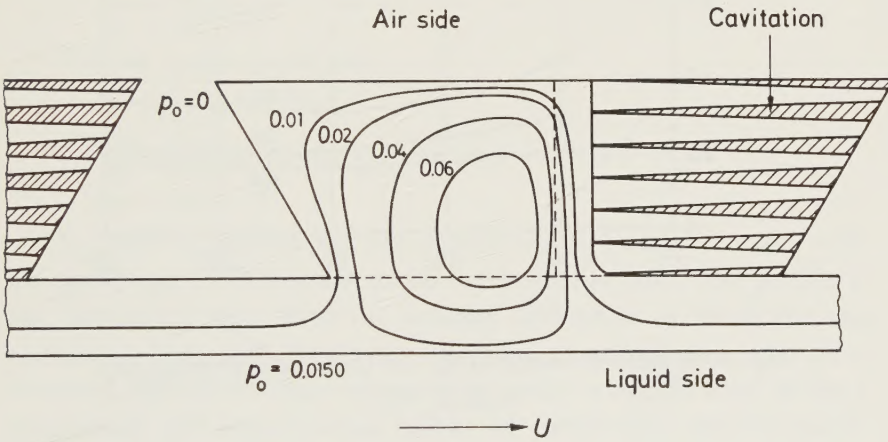


Fig. 4. Pressure field for a bidirectional seal with $a_o=0.260$, $b_o=0.1$, $c_o=0.6$, and $\alpha=60^\circ$. $\Delta p_o=0.0150$.

From the pressure fields are obtained the radial load capacities, the liquid flows, and the power losses.

The seal should be able to maintain a certain pressure difference without leakage. Therefore the relative power loss Φ , defined as total power loss per unit pressure difference, is introduced. The contact surface is then said to be optimum when Φ is as small as possible. It was found that for both unidirectional and bidirectional seals b_o should be as small as possible. In figs. 5 and 6 minimum values of the non-dimensional relative power loss Φ_o are shown for $b_o = 0.1$ which is the smallest value of b_o that was used in the calculations.

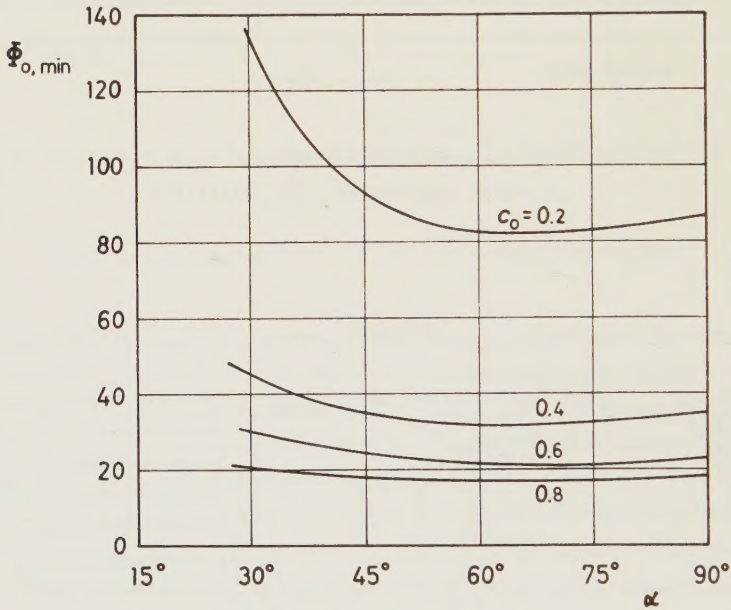


Fig. 5. Minimum values of Φ_o for the unidirectional seal with $b_o = 0.1$

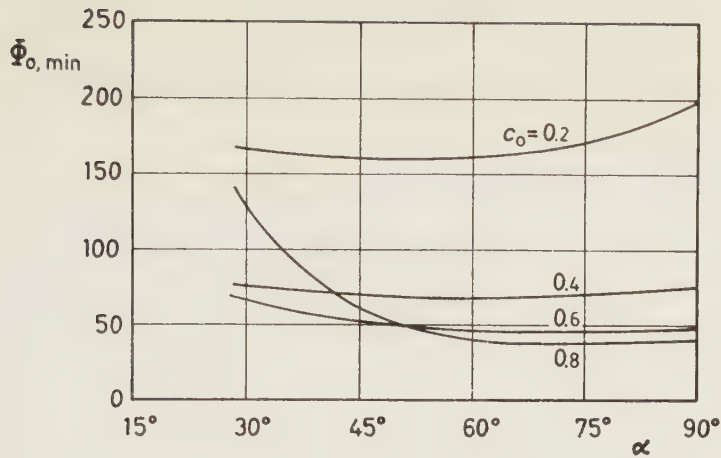


Fig. 6. Minimum values of Φ_0 for the bidirectional seal
with $b_0 = 0.1$

From these figures the optimum contact geometry is obtained.
For the unidirectional seal:

$$a_0 = 0.4 - 0.5$$

$$b_0 = 0.1$$

$$c_0 = 0.8$$

$$\alpha = 60^\circ - 75^\circ$$

For the bidirectional seal:

$$a_0 = a_{0 \max} = 0.17$$

$$b_0 = 0.1$$

$$c_0 = 0.8$$

$$\alpha = 60^\circ - 75^\circ$$

In order to control the theoretical values, tests were made with an optimum unidirectional metal seal with the same film profile as the theoretical one. Both the pressure difference, at which the seal started to leak, and the power loss were measured. The agreement between theoretical and experimental values is satisfactory. In fig. 7 theoretical and experimental values of power loss are compared.

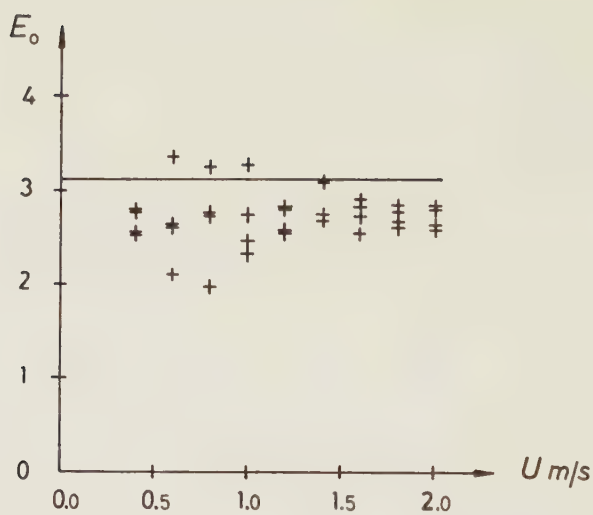


Fig. 7. Theoretical curve and test points of E_o for the unidirectional metal test seal

Calculation of Contact Force and Optimization

with Regard to the Seal Cross Section

In papers by Horve (1) and Schmitt (2) the contact force is calculated in different analytical ways but they both transform the actual cross section into a simplified one. Here the finite element method is used and the cross section can be treated without simplification. Since the static case is considered and the strains are small the viscoelastic properties of the seal material can be neglected and sufficiently accurate results are obtained with ordinary linear theory of elasticity. The cross section has the same form as the one used by a Swedish seal manufacturer and is close to the optimum cross section for radial sharp lip seals proposed by Symons (3). In fig. 8 the deformation of the cross section is shown.

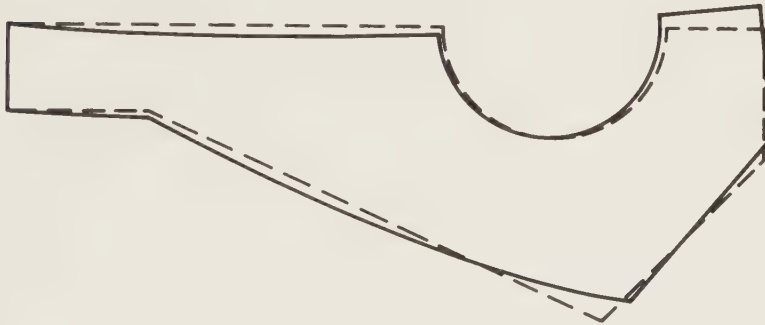


Fig. 8. Deformation of the cross section for the case $R_o=5$, $\Delta R_o=0.025$, $F_{os}=0.05$, and $\Delta p_{oc}=0$. The dotted line indicates the initial shape.

It is now possible to optimize the seal when consideration is taken to the fact that the lubricating film must be able to carry the imposed load. For the unidirectional seal the optimum contact geometry is found to be the same as mentioned above. For the bi-directional seal the following optimum values are found:

$$a_o \doteq a_{o \max} = 0.35$$

$$b_o = 0.1$$

$$c_o = 0.6$$

$$\alpha = 60^\circ - 75^\circ$$

Tests were made with both unidirectional and bidirectional standard seals. The power loss was measured at different pressure differences and shaft speeds and the contact surface was photographed. In figs. 9 and 10 theoretical and experimental values of power loss are shown.

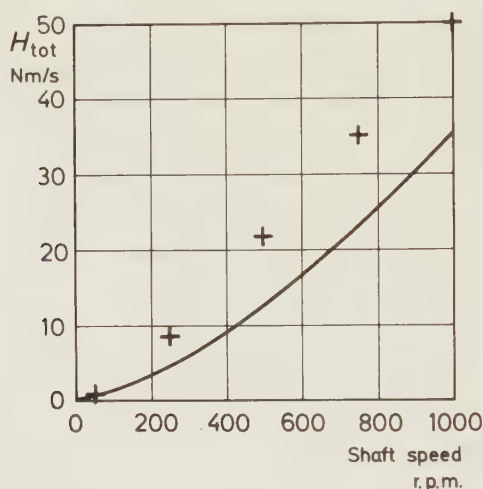


Fig. 9. Theoretical curve and experimental points of power loss for the unidirectional test seal at $\Delta p = 0.5$ bar

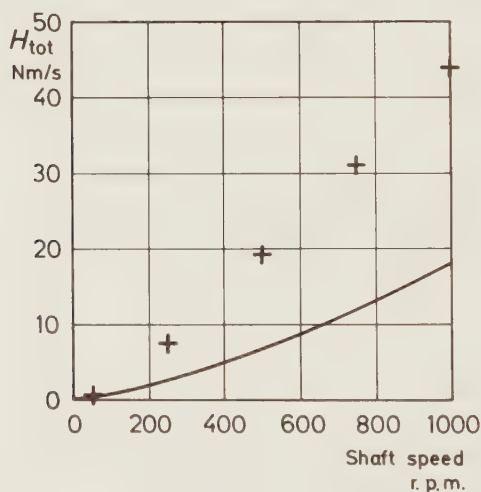


Fig. 10. Theoretical curve and experimental points of power loss for the bidirectional test seal at $\Delta p = 0.5$ bar

The standard test seals have film profiles that certainly deviate from the optimum theoretical film profile. Taking this into consideration the agreement between theoretical and experimental power loss is satisfactory.

References

1. Horve, L.A.: The calculation of shaft seal steady state radial loads with non-uniform cross-section and local stretch forces included. Trans. ASLE 13, 288-294 (1970).
2. Schmitt, W.A.: Radial load as a lip seal design and quality control factor. Trans. ASME 90, 405-411 (1968).
3. Symons, J.D.: Optimum lip design by fractional factorial experimentation. SAE paper 660380 (1966).

